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OVERALL PERFORMANCE OF 6-INCH RADIAL-BLADED CENTRIFUGAL COMPRESSOR WITH VARIOUS DIFFUSER VANE SETTING ANGLES

by Calvin L. Ball, Carl Weigel, Jr., and Edward R. Tysl

Lewis Research Center
Cleveland, Ohio 44135

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16. Abstract

Overall compressor performance is presented as a function of equivalent weight flow for five different diffuser vane setting angles. For design speed, the weight flow corresponding to the peak efficiency was varied from 100 percent of design flow to 65 percent by operating the compressor with the five different diffuser vane angles. Over the range of diffuser vane setting angles tested, the peak efficiency of the compressor remained essentially constant at 78 percent.

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SUMMARY

A 6-inch (15.2-cm) diameter radial-bladed centrifugal compressor applicable for a 10-kilowatt Brayton-cycle space power system was tested with five different diffuser vane setting angles. The tests were conducted using argon gas at an inlet pressure of 6 psia (4.1 N/cm^2 abs). Overall performance is presented as a function of equivalent weight flow for 60 and 100 percent of design speed.

For design speed, the weight flow corresponding to the peak efficiency was varied from 100 percent of design flow to 65 percent by operating the compressor with the five different diffuser vane angles. Over the range of diffuser vane setting angles tested, the peak efficiency of the compressor for design speed remained essentially constant at 78 percent.

Analysis of the overall performance data indicates that the flow range of this compressor (maximum flow and surge flow) is controlled primarily by the diffuser vanes for the range of diffuser vane setting angles tested. This reflects a broad flow range for the impeller of this compressor as compared to its diffuser vanes. However, measured wall-static-pressure gradients at the impeller inlet and through the diffuser vane section indicate that with the diffuser vane set at 70.32° (high flow setting) the impeller also may be approaching a limiting flow. Therefore, because of the influence of the impeller in restricting the flow, diffuser vanes set at lower angles than 70.32° (higher flow settings) may not necessarily result in appreciably higher compressor flows.

INTRODUCTION

One of the methods which shows promise for generating electrical power in space is the closed Brayton-cycle system. For different space missions the required amounts of electrical power will vary. Because of the time and cost involved in developing the rotating machinery for the Brayton-cycle system, it is desirable to take a basic set of hardware developed for a nominal power level and utilize it for a range of power outputs. This can be accomplished by varying the mass flow rate. The mass flow rate can be adjusted in two ways: by varying the volume flow rate, and by adjusting system pressure level to vary the gas density.

To achieve an appreciable change in volume flow rate requires the turbomachinery blading to be rematched for the new throughflow velocities and resulting flow angles. Varying the gas density by adjusting system pressure level results in a change in Reynolds number. Both of these methods can result in changes in compressor efficiency. To obtain the best efficiency, it may be desirable or necessary to adjust both volume flow rate and system pressure level.

In order to optimize compressor efficiency for a specified power level, more experimental data are required on machines typical of those considered for the Brayton-cycle space power systems. To provide additional data on the effects of rematching compressors to different volume flows and to better establish the change in performance with Reynolds number, an experimental program was initiated at Lewis using a small radial-bladed centrifugal compressor. The overall performance of this compressor was described in reference 1. The effect of Reynolds number on performance was reported in reference 2. The design and fabrication of this compressor was accomplished by contract to the AiResearch Manufacturing Company, Division of the Garrett Corporation, Phoenix, Arizona, and is presented in reference 3. The compressor was designed as part of a two-shaft, Brayton-cycle space power system having a design electrical power output of 10 kilowatts.

This report presents the results of a study to determine the effectiveness of resetting the diffuser vanes of this compressor to different angles as a means of adjusting the volume flow rates over which the compressor operates, and to establish the resulting effect on compressor peak efficiency. The compressor was tested in argon gas with five different diffuser vane setting angles. Overall performance for the compressor with the five different diffuser vane angles is presented as a function of equivalent weight flow. Plots of static-pressure rise through the compressor with the various diffuser vane setting angles are also presented and a comparison with the design static-pressure rise is made. The compressor inlet pressure was maintained at the design value of 6 psia (4.1 N/cm^2 abs) and an inlet temperature of 536° R (298 K) throughout the tests. The performance is presented for 60 and 100 percent of design speed. All performance data reported herein were obtained from tests conducted at Lewis.

COMPRESSOR DESIGN

The compressor was designed for argon gas as the working fluid. The values of the

TABLE I. - COMPRESSOR DESIGN PARAMETERS

[Working fluid, argon.]

Compressor design parameters	Based on design inlet pressure and temperature	Based on standard inlet pressure and temperature
Inlet total pressure, P_1, psia; N/cm^2 abs	6; 4.1	14.7; 10.1
Inlet total temperature, T_1, ^{o}R; K	536; 298	518.7; 288
Weight flow rate, W, lb/sec; kg/sec	0.611; 0.278	1.52; 0.69
Compressor total-pressure ratio, P_6/P_1	2.38	[a]2.38
Compressor total-temperature ratio, T_6/T_1	1.525	[a]1.525
Compressor efficiency, η_{1-6}	0.798	[a]0.798
Impeller total-pressure ratio, P_3/P_1	2.62	[a]2.62
Impeller efficiency, η_{1-3}	0.896	[a]0.896
Rotative speed, rpm	38 500	37 900
Impeller tip speed, U_{t3}, ft/sec; m/sec	1004; 306	989; 302
Impeller slip factor, f_s	0.830	[a]0.830
Impeller windage factor, f_w	0.039	[a]0.039
Compressor work factor, f_{cw}	0.869	[a]0.869
Specific speed, N_s	0.1057	0.1057
Reynolds number, Re	1.3×10^6	3.5×10^6

[a]These values are approximate and in actuality will differ from design as a result of the difference in Reynolds number.

compressor design parameters and the equivalent values for standard inlet conditions are given in table I. A detailed description of the design and of the procedure employed in arriving at the final design point values is given in reference 3. A summary of the design is presented in reference 1. The diffuser vane setting angles for which the compressor was to be tested, and their respective design weight flows, are presented in table II. Percent of compressor design equivalent weight flow for the different diffuser vanes is also given in table II.

The leading edge of the diffuser vanes was maintained at a fixed radial position with respect to the impeller discharge and thus the short radial vaneless diffuser section remained the same for the different diffuser vane setting angles tested. The chord length and vane shape were also the same for all diffusers.

The diffuser vane set at 75.75^{o} as measured from the radial direction is the design diffuser vane setting angle. As mentioned in reference 1, this diffuser vane setting angle resulted in lower than design weight flow. The diffuser vane having the setting angle of 72.75^{o} was employed in obtaining the data presented in reference 1. The overall performance of the compressor with this diffuser vane setting angle more closely approached design conditions.

TABLE II. - CALCULATED DESIGN EQUIVALENT WEIGHT FLOW FOR EACH OF THE FIVE DIFFERENT DIFFUSER VANE SETTING ANGLES AT WHICH THE COMPRESSOR WAS TESTED

Diffuser vane setting angle, deg	Design equivalent weight flow		Percent of compressor design equivalent weight flow
	lb/sec	kg/sec	
80.75	1.22	0.55	80
78.75	1.34	.61	88
[a]75.75	1.52	.69	100
72.75	1.70	.77	111
70.32	1.84	.83	121

[a]Design diffuser vane setting angle.

APPARATUS AND PROCEDURE

The test apparatus, test facility, instrumentation, and test procedure are the same as that reported in reference 1, with the exception of the different diffuser vane ring assemblies.

The impeller is shown in figure 1 with one of the diffuser vane rings. The five diffuser vane rings with the vane setting angles which were tested in combination with the impeller are shown in figure 2. The diffusers have 23 vanes which are integral with the vaned diffuser blade ring. The diffuser vane tips were contoured to fit the machined aluminum casting that comprised the compressor scroll, impeller shroud, and inlet. This casting is shown in figure 3 as part of the centrifugal compressor assembly.

Figure 1. - 6-Inch (15.2-cm) radial-bladed compressor impeller and vaned diffuser.

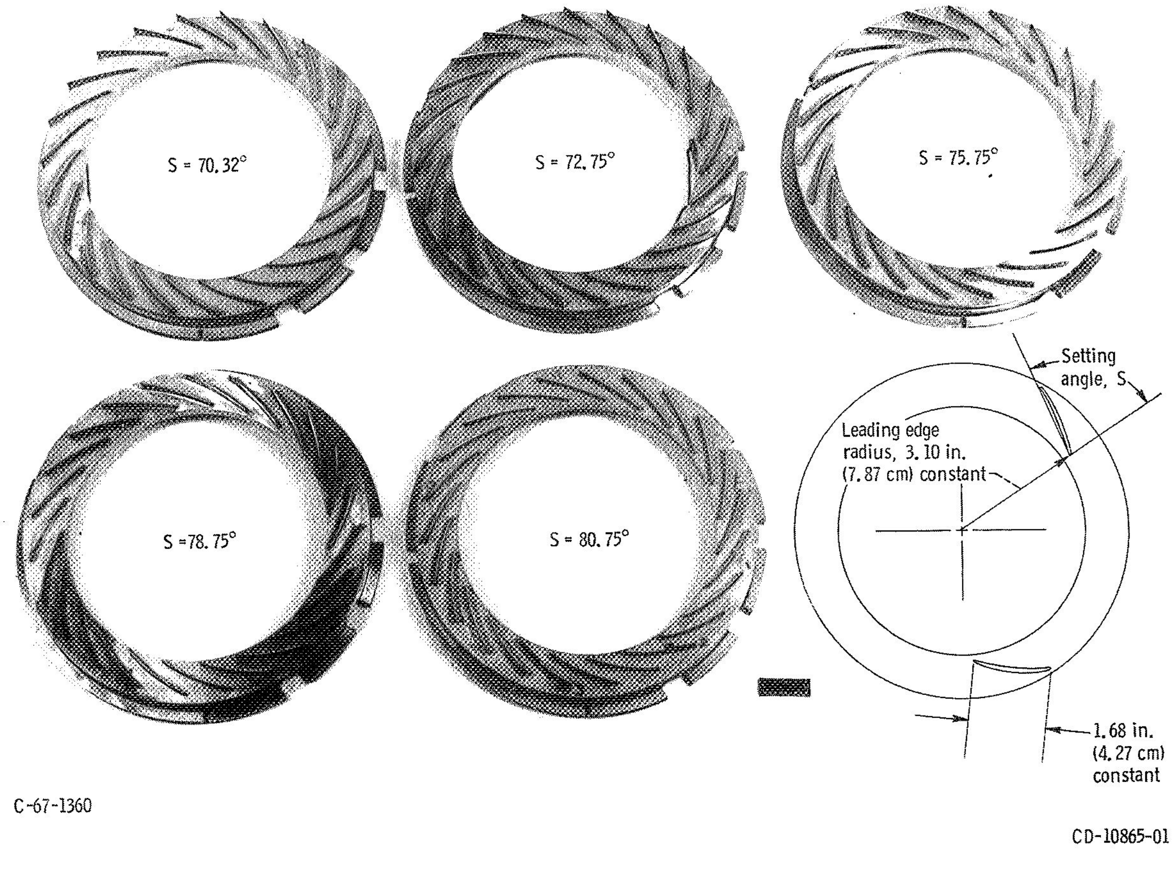


Figure 2. - Compressor vaned diffusers with various diffuser vane setting angles.

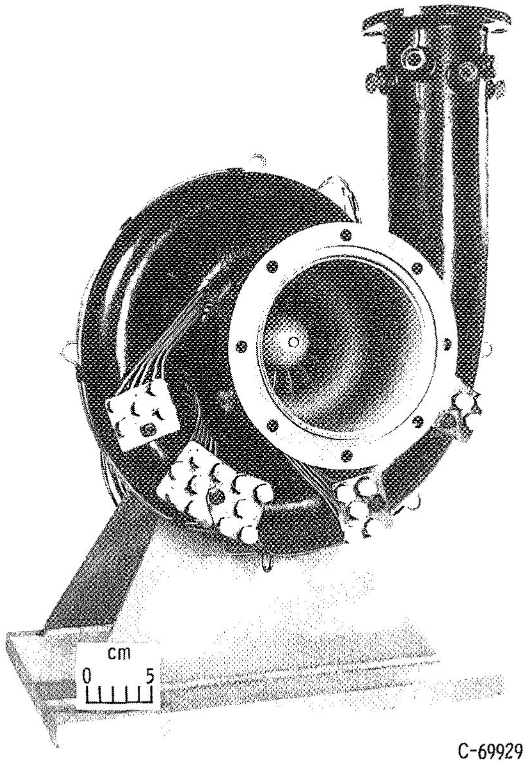


Figure 3. - Centrifugal compressor assembly.

A cutaway view of the compressor assembly is shown in figure 4 with the locations of instrumentation and calculation stations. A cross-sectional view of the instrumentation of stations 1 and 6 is also shown. At station 2, five wall-static-pressure taps were located at equal distances of 0.19 inch (0.48 cm) along the shroud over the impeller inlet. Between stations 4 and 5, six wall-static-pressure taps were located at equal distances along the diffuser vane midpassage, with the exception of the 70.32° diffuser vane, which has only five wall taps.

The static (cold) impeller clearance between the shroud and blade tip at the impeller discharge was set at 0.010 inch (0.025 cm) for all diffuser vane tests.

Compressor test data were taken over a range of weight flows from maximum flow to surge conditions for 60 and 100 percent of equivalent design speed. Throughout the test series, inlet pressure and inlet temperature were maintained at nominal design conditions of 6 psia (4.1 N/cm^2 abs) and 536° R (298 K).

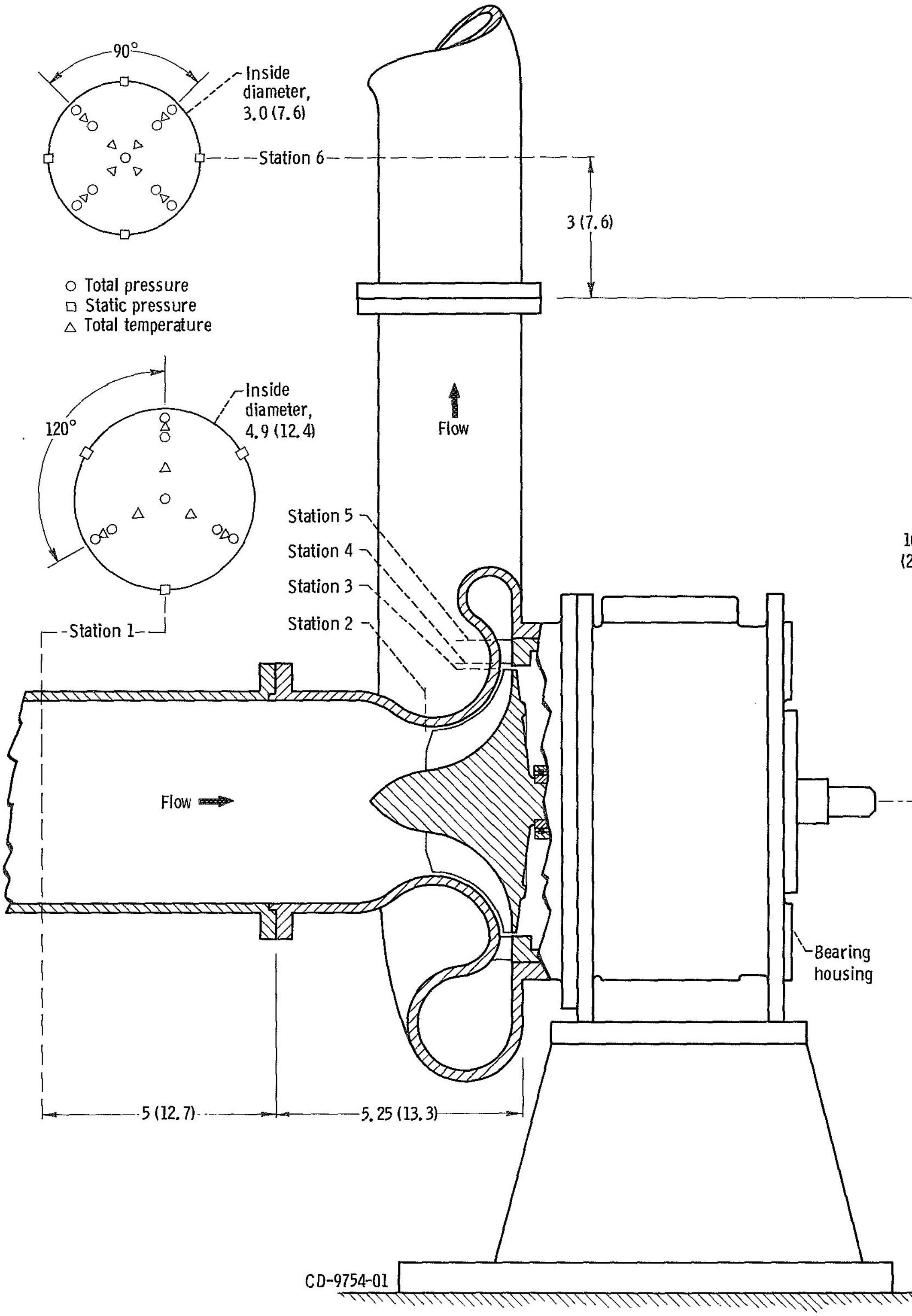


Figure 4. - Compressor cross section, showing instrument locations. (All dimensions are in inches (cm).)

RESULTS AND DISCUSSION

The data presented herein were obtained from a 6-inch (15.2-cm) diameter radial-bladed centrifugal compressor operated with argon as the working fluid. The impeller was tested in combination with five different diffuser vane setting angles.

The overall performance of the compressor with the five different vane setting angles is presented in figure 5. Total-pressure ratio, compressor work factor, and adiabatic efficiency are presented as a function of argon weight flow for 60 and 100 percent of design speed. For ease of comparing the effect of diffuser vane setting angle on the pressure ratio, work factor, and efficiency, these parameters are presented as a percent of design equivalent weight flow [1.52 lb/sec (0.69 kg/sec)] for the five diffuser vane setting angles in figure 6 for design speed. (The trends in performance with respect to diffuser vane setting angle at 60 percent of design speed are the same as those for 100 percent design speed and therefore are not included in the comparison plot.) As can be seen in figure 6, the peak efficiency remained essentially constant at 0.78 for the compressor over the range of diffuser vane setting angles tested. The peak efficiencies for the compressor with the diffuser vane angle set at 70.32° (high flow setting) occurred at approximately design flow. And with the diffuser vane set at 80.75° the peak efficiency for the compressor occurred at approximately 65 percent of design flow. The weight flow at which peak efficiency occurred for the compressor with the various diffuser vane setting angles was considerably (~20 percent) below the design flows for the diffuser vanes noted in table II. As indicated in reference 1 and mentioned in the design section of this report, the design diffuser vane setting angle, 75.75°, resulted in a weight flow that was too low and the diffuser vane set at 72.75° was used to more closely achieve design performance. This decrease in flow range was attributed to the actual boundary-layer blockage being greater than design, which caused the design diffuser vane to operate closer to a flow choking condition than was assumed in the design. Opening up the diffuser vanes to a setting angle of 72.75° resulted in the vanes operating closer to design incidence and thus the compressor more closely approaching design performance.

The change in overall flow range achieved by resetting the diffuser vanes to different angles (fig. 6) was close to that which was calculated (table II).

It would appear that the diffuser vanes over the range of setting angles tested are the primary component in limiting compressor flow range, with the impeller having only a secondary effect. Thus, the flow range of the impeller must be considerably greater than that of a given diffuser.

Compressor work factor and compressor pressure ratio corresponding to peak efficiency (fig. 6) tended to increase slightly as the diffuser vane setting angle was changed from 70.32° to 72.75°. As the diffuser vane setting angle was further reduced, these parameters tended to decrease slightly. The fact that the work factor for the compressor

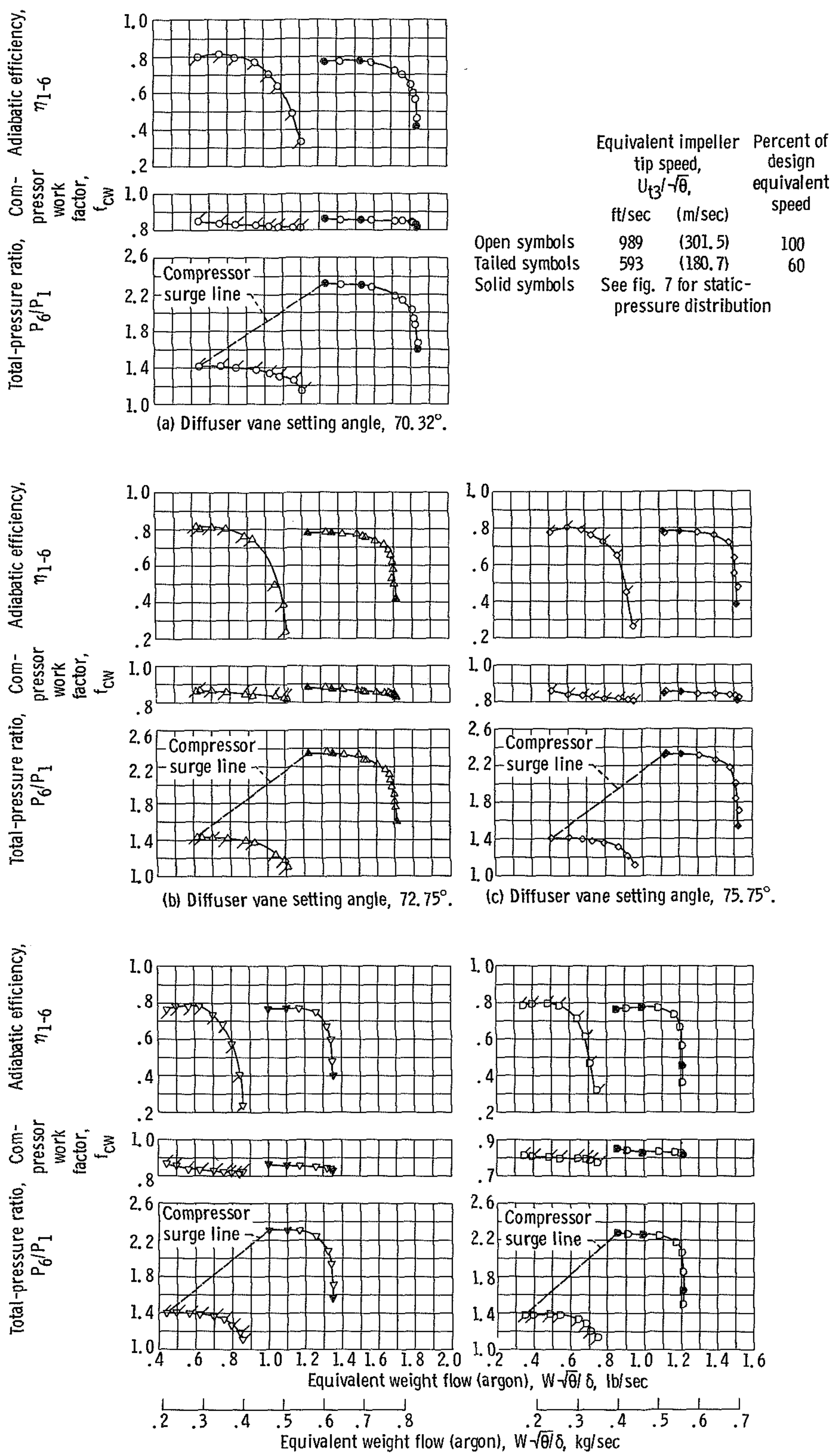


Figure 5. - Overall performance of 6-inch (15.2-cm) radial-bladed centrifugal compressor.

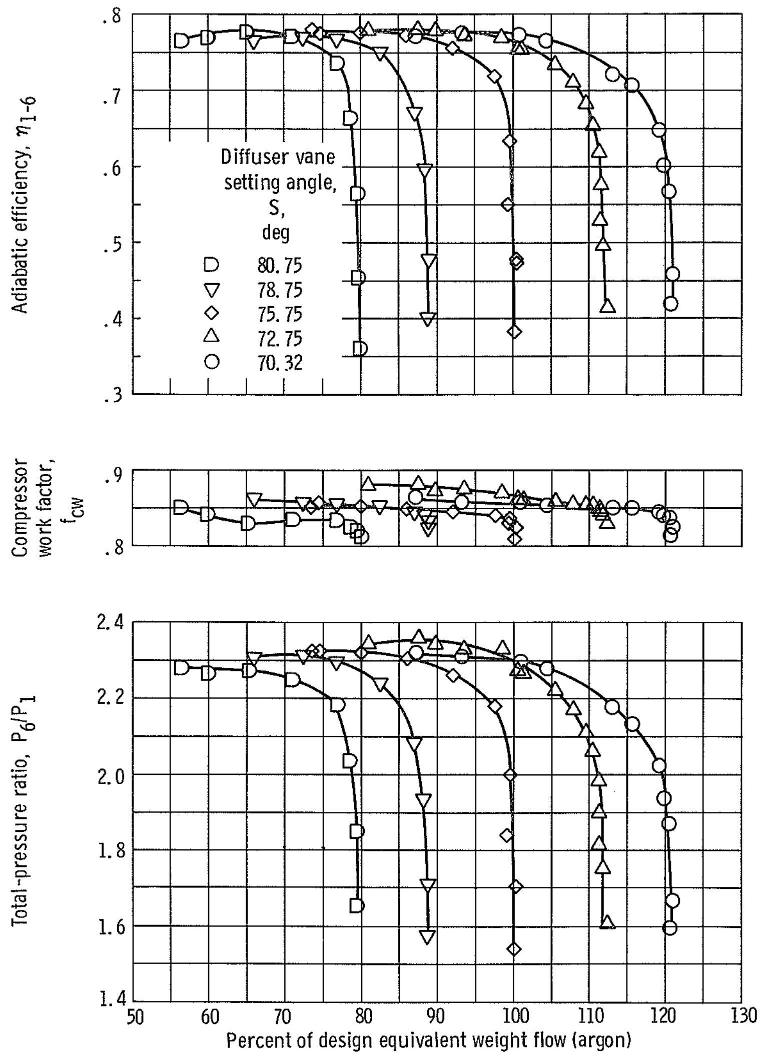


Figure 6. - Comparison of overall performance of a 6-inch (15.2-cm) radial-bladed centrifugal compressor at design speed, with five different diffuser vane setting angles.

with the various diffuser vanes does not fall on one line indicates that this parameter, which one might assume to be only a function of the impeller design, flow, and speed, is also affected by the diffuser vanes. This may, in part, be attributed to how the diffuser vanes affect the flow leaving the impeller and thus to the energy which the impeller imparts to the fluid.

The flow range for the compressor with a given diffuser vane tends to decrease (as a percent of compressor design flow) with increasing vane setting angle. However, the ratio of maximum flow for the compressor with a given diffuser vane to that at which surge occurs with that diffuser vane remains essentially constant over the range of diffuser vane setting angles tested. For a given change in weight flow, the change in incidence angle decreases as the diffuser vane setting angle is increased. Therefore, if it were assumed that maximum flow and surge flow were controlled only by the magnitude of the change in incidence angle with respect to design incidence, and independently of diffuser vane angle, a broader flow range would be expected at the higher setting angles. However, other factors such as increased vane metal blockage (as a percent of flow area) and increased vane aerodynamic loading at the higher vane setting angles no doubt tend to limit the flow range, as apparent from the data presented in figure 6. This trend towards decreasing flow range (incidence angle range) as the blade setting angle is increased has also been observed in other test data (refs. 4 and 5).

The static-pressure rise through the compressor for design speed with the five different diffuser vane setting angles was measured and is presented in figure 7. The data are presented for three weight flows for each of the diffuser vanes: one near maximum flow, one near peak efficiency, and one near compressor surge. The associated overall performance points are indicated by the solid symbols in figure 5. Figure 7(a) presents the data obtained from the diffuser vanes set for the high flow. Figures 7(b) to (e) present the data obtained from the diffuser vanes set for progressively lower flows. Also presented in figure 7 as a reference is the design static-pressure rise for the compressor.

The static-pressure gradients presented in figure 7 indicate that a high velocity exists at the diffuser vane inlet at the high flow condition for all diffuser vane setting angles. This is apparent from the rapid drop in static pressure in the diffuser vane passage. Following this high velocity there appears to be a flow shock, which is indicated by the rapid rise in static pressure. The presence of the flow shock appears to have severely affected the performance of the diffuser vanes. This, no doubt, is a prime factor in limiting the high-flow range of the compressor. The compressor flow rate at which this condition appears in the diffuser vane should be primarily a function of the flow incidence angle on the diffuser vane. If this were the only factor affecting the maximum compressor flow, it could be controlled by adjustment of the diffuser vane setting angle. However, at the high flow rates associated with the lower diffuser vane setting angles, high velocities are also starting to appear at the impeller leading edge. This is attributed to the lower incidence angles at which the impeller is operating at the higher flow rates.

Since the maximum flow of the compressor responded to the diffuser vane adjustment, as would be expected from their differences in design flow, it is felt that the high velocities that were starting to appear at the impeller inlet at the high flows were not seriously affecting the flow range of the compressor. However, diffuser vanes set for higher flows may not result in appreciably higher compressor flows because of the impeller controlling the maximum flow.

Compressor surge also appears to be controlled by the diffuser vanes (based on the

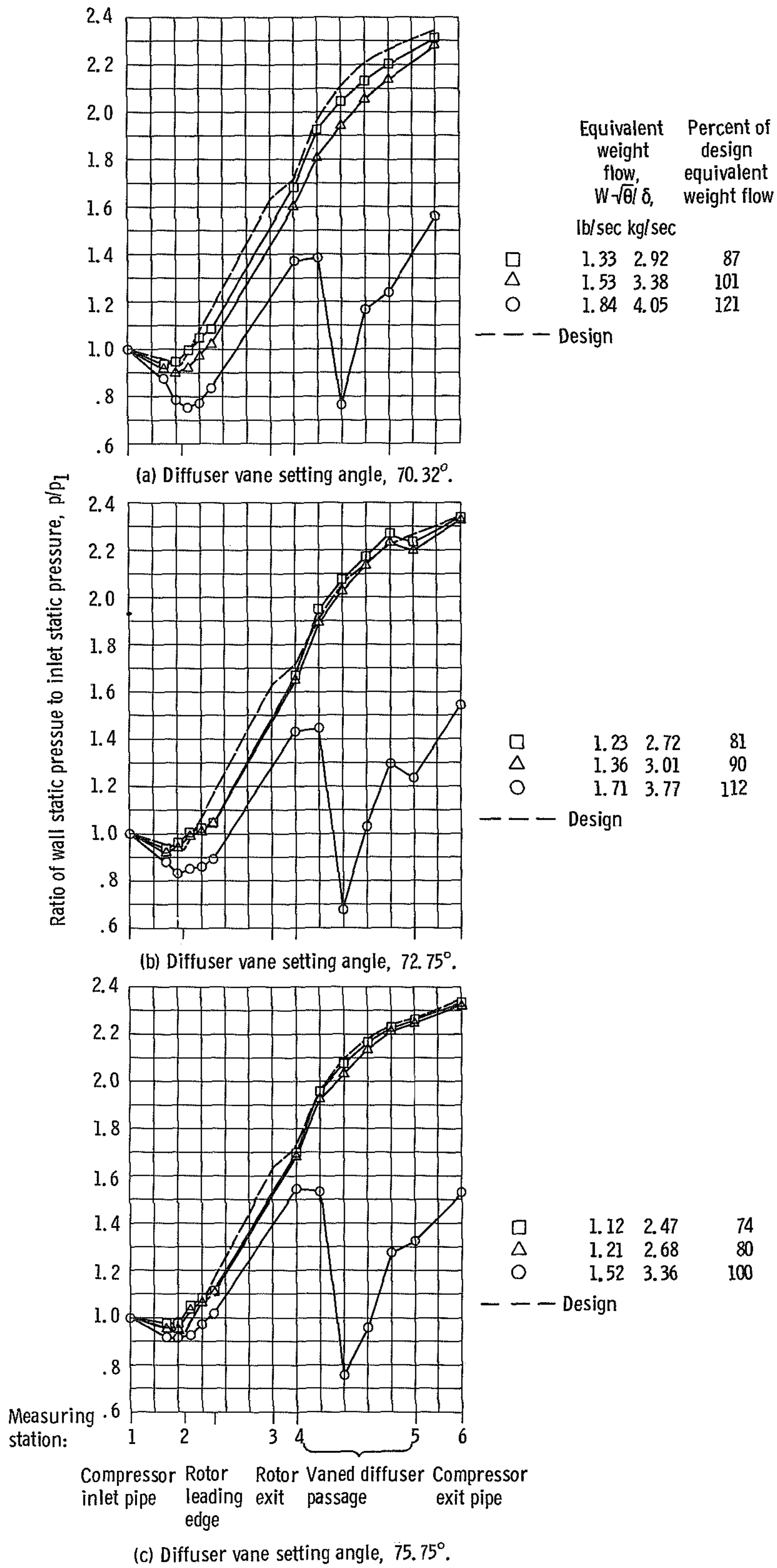
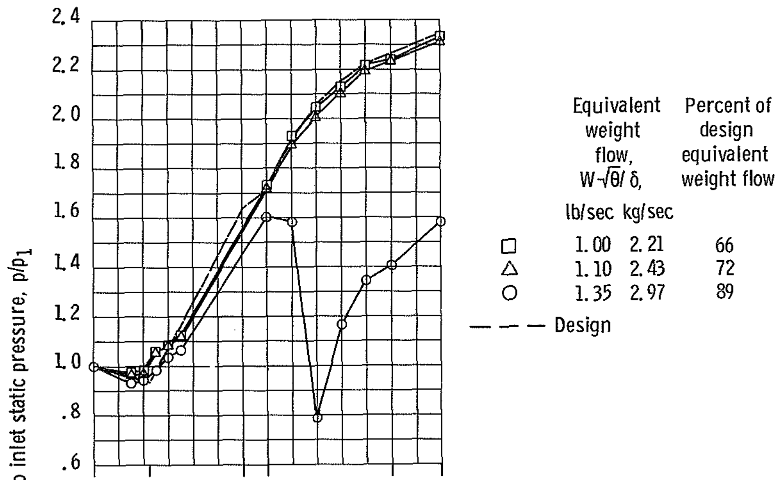
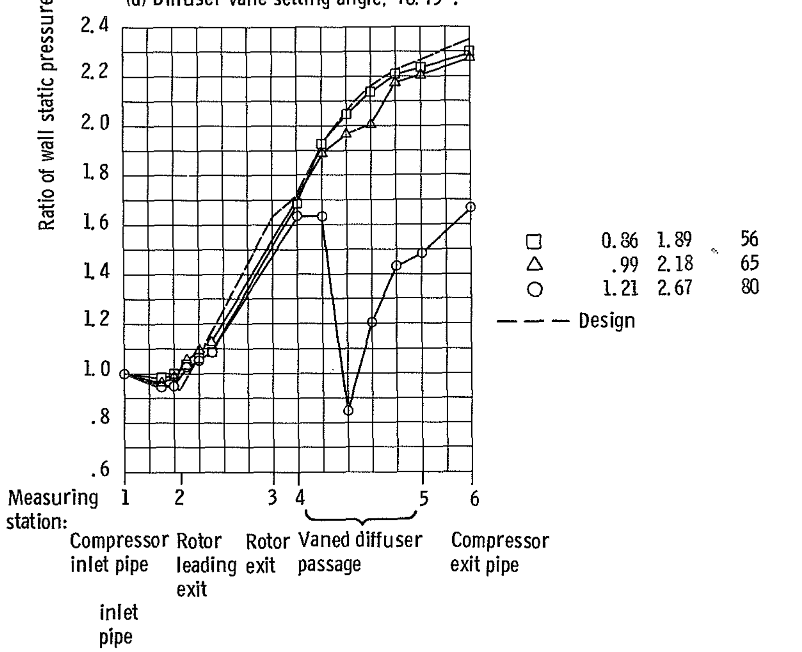


Figure 7. - Compressor static-pressure distribution at design speed for three weight flows.



(d) Diffuser vane setting angle, 78.75°.



(e) Diffuser vane setting angle, 80.75°.

Figure 7. - Concluded.

overall performance). However, there is no significant difference in static-pressure rise through the diffuser vane section (based on wall taps located at midvane passage) as surge is approached to reflect a stalling of the diffuser vanes which would precipitate surge.

An appreciable flow-range adjustment was achieved for this compressor by resetting its diffuser vanes to different angles. Also the peak efficiency of the compressor for the range of diffuser vane setting angles tested remained essentially constant. Therefore, it can be concluded that varying the power level in a Brayton-cycle electrical power generation system by adjusting volume flow is an acceptable method with this compressor within the flow range adjustment demonstrated, requiring only a change in diffuser vane setting angle. As noted in reference 2, a progressive degradation in peak efficiency along with a lowering of the flow at which it occurred was observed with decreasing system pressure level or decreasing Reynolds number for this compressor. This degradation in efficiency was more pronounced at the lower levels of Reynolds number. From the observed effects of volume flow and system pressure level adjustment on this compressor's performance, it can be concluded that the optimum approach in terms of achieving maximum efficiency for this compressor over a broad range of system power levels may require a combination of both volume flow and system pressure level adjustment.

CONCLUDING REMARKS

A 6-inch (15.2-cm) radial-bladed centrifugal compressor was tested with five different diffuser vane setting angles. The tests were conducted using argon gas. The data presented include overall performance and measured static-pressure rises through the compressor with the various diffuser vanes. For design speed, the weight flow corresponding to the peak efficency was varied from 100 percent of design flow to 65 percent by operating the compressor with the five different diffuser vane angles. Over the range of diffuser vane setting angles tested, the peak efficiency of the compressor for design speed remained essentially constant at 78 percent.

Analysis of the overall performance data indicates that the flow range of this compressor (from maximum flow to compressor surge) is primarily controlled by the diffuser vanes for the range of diffuser vane setting angles which were tested. This reflects a broad flow range for the impeller of this compressor as compared to its diffuser vanes. However, measured wall-static-pressure gradients at the impeller inlet and through the diffuser vane section indicate that with the diffuser vane set at 70.32° (high flow setting) the impeller also may be approaching a limiting flow. Therefore, diffuser vanes set at lower angles than 70.32° may not necessarily result in higher compressor flows because of the influence of the impeller in restricting the flow.

It is also apparent that the compressor work factor is not solely a function of impeller design, rotational speed, and compressor flow rate but also is affected by the diffuser vanes. This is apparently a result of the manner in which the diffuser vanes affect the flow patterns or gradients within the impeller.

From the observed effects of volume flow and system pressure level adjustment on this compressor's performance, it can be concluded that the optimum approach in terms of achieving maximum efficiency for this compressor over a broad range of system power levels may require a combination of both volume flow and system pressure level adjustment.

Lewis Research Center,
National Aeronautics and Space Administration,
Cleveland, Ohio, August 12, 1970,
120-27.

APPENDIX - SYMBOLS

c diffuser vane chord, 1.68 in.; 4.27 cm

c_p specific heat at constant pressure of argon, 0.125 Btu/(lb)(^oR); 524 J/(kg)(K)

D diameter, ft; m

f_{cw} compressor work factor, $gJc_p(T_6 - T_1)/U_{t3}^2$

f_s slip factor, $V_{ut3}/U_{t3} = gJc_p \Delta T_{vd}/U_{t3}^2$

f_w windage factor, $gJc_p \Delta T_w/U_{t3}^2$

g gravitational acceleration, 32.17 ft/sec^2; 9.807 m/sec^2

ΔH_{is} isentropic specific work, (ft)(lb)/lb; (m)(N)/kg

J mechanical equivalent of heat, 778.16 (ft)(lb)/Btu; 1.00 (m)(N)/J

N_s specific speed, $RPM\sqrt{Q}/60(g\Delta H_{is})^{3/4}$

P total (stagnation) pressure, psia; N/cm^2 abs

p static pressure, psia; N/cm^2 abs

Q volume flow, ft^3/sec; m^3/sec

R gas constant (argon), 38.683 (ft)(lb)/(lb)(^oR); 208.13 (m)(N)/(kg)(K)

Re Reynolds number, $\rho_1 U_{t3} D_{t3}/\mu_1$

RPM impeller rotational speed, rpm

S diffuser vane setting angle, deg

T total (stagnation) temperature, ^oR; K

ΔT_{vd} gas temperature rise associated with vector diagrams, ^oR; K

ΔT_w gas temperature rise associated with windage, ^oR; K

U impeller wheel speed, ft/sec; m/sec

V absolute gas velocity, ft/sec; m/sec

W weight (mass) flow rate, lb/sec; kg/sec

γ ratio of specific heat at constant pressure to specific heat at constant volume (for argon, 1.667)

δ ratio of compressor inlet total pressure to NASA standard sea-level pressure, $P_1/14.7$ psia; $P_1/10.1$ N/cm^2 abs

η adiabatic temperature rise efficiency, $T_1\left[(P_6/P_1)^{(\gamma-1)/\gamma} - 1\right]/(T_6 - T_1)$

θ ratio of compressor inlet total temperature to NASA standard sea-level temperature, $T_1/518.7^o$ R; $T_1/288.2$ K

μ dynamic viscosity, 1.49×10^{-5} lb/(sec)(ft) at 518.7^o R; 2.22×10^{-5} (N)(sec)/m^2 at 288.2 K

ρ density, 0.1055 lb/ft^3 at 518.7^o R and 14.7 psia; 1.690 kg/m^3 at 288.2 K and 10.1 N/cm^2 abs

Subscripts:

is isentropic

t tip

u tangential component

1 station in inlet pipe 5 in. (12.7 cm) upstream of compressor inlet flange (fig. 4)

2 station at impeller inlet (fig. 4)

3 station at impeller outlet (fig. 4)

4 station at diffuser blade inlet (fig. 4)

5 station at diffuser blade outlet (fig. 4)

6 station in exit pipe 3 in. (7.6 cm) downstream of compressor scroll exit flange (fig. 4)

REFERENCES

1. Tysl, Edward R.; Ball, Calvin L.; Weigel, Carl; and Heidelberg, Laurence J.: Overall Performance in Argon of a 6-Inch Radial-Bladed Centrifugal Compressor. NASA TM X-1622, 1968.

2. Heidelberg, Laurence J.; Ball, Calvin L.; and Weigel, Carl: Effect of Reynolds Number on Overall Performance of a 6-Inch Radial Bladed Centrifugal Compressor. NASA TN D-5761, 1970.

3. Anon.: Design and Fabrication of a High-Performance Brayton-Cycle Compressor Research Package. Rep. APS-5109-R, AiResearch Mfg. Co. (NASA CR-54368), May 1965.

4. Taylor, W. E.; Murrin, T. A.; and Colombo, R. M.: Systematic Two-Dimensional Cascade Tests. Vol. I: Double Circular-Arc Hydrofoils. Rep. VARL-H910254-50, United Aircraft Corp. (NASA CR-72498), Dec. 19, 1969.

5. Emery, James C.; Herrig, L. Joseph; Erwin, John R.; and Felix, A. Richard: Systematic Two-Dimensional Cascade Tests of NACA 65-Series Compressor Blades at Low Speeds. NACA Rep. 1368, 1958.

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